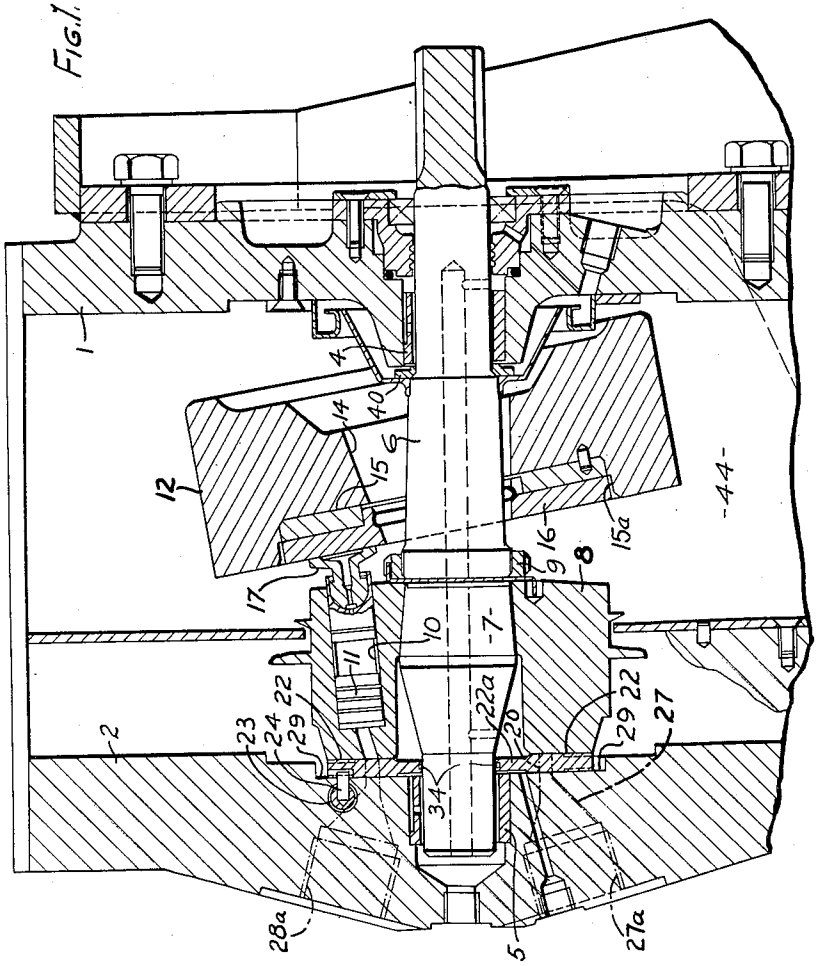
May 14, 1963 D. FIRTH ET AL 3,089,427

HYDRAULIC MACHINES

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FIG. 1.



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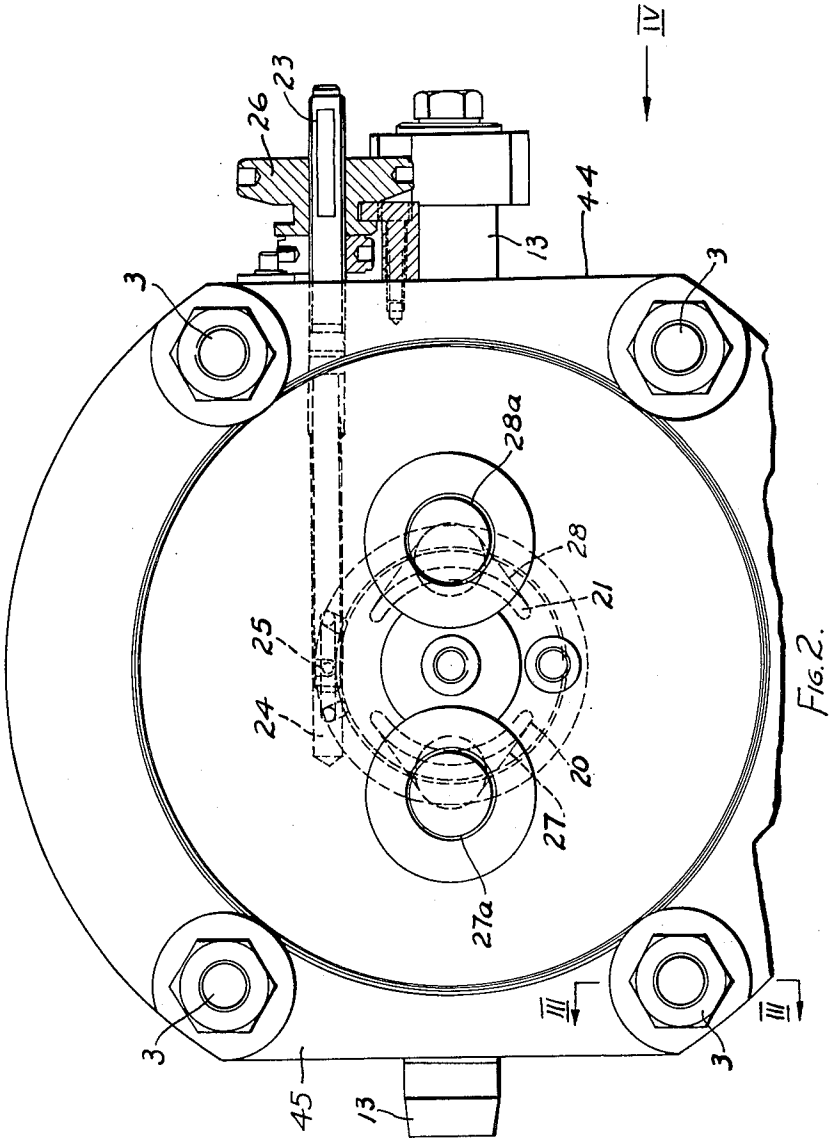


FIG. 2.

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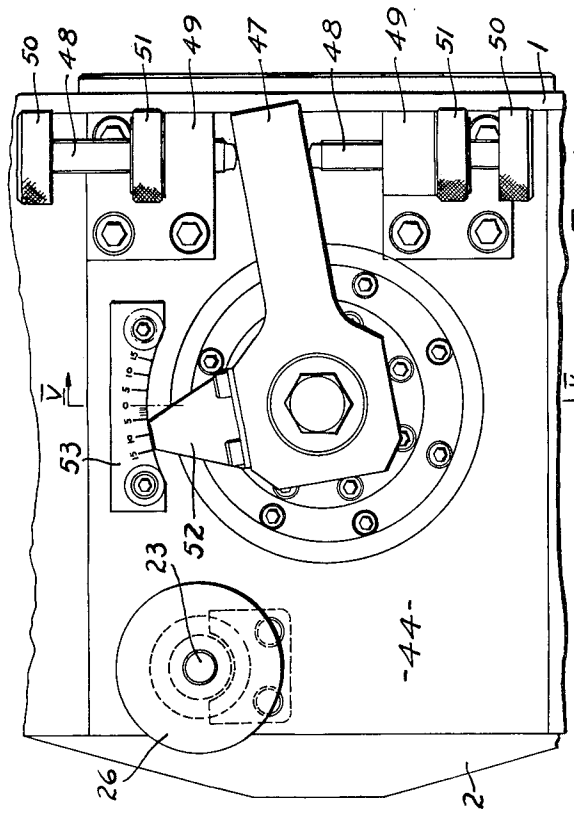


FIG. 4.

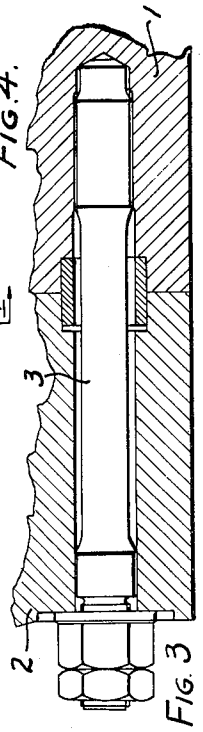


FIG. 3

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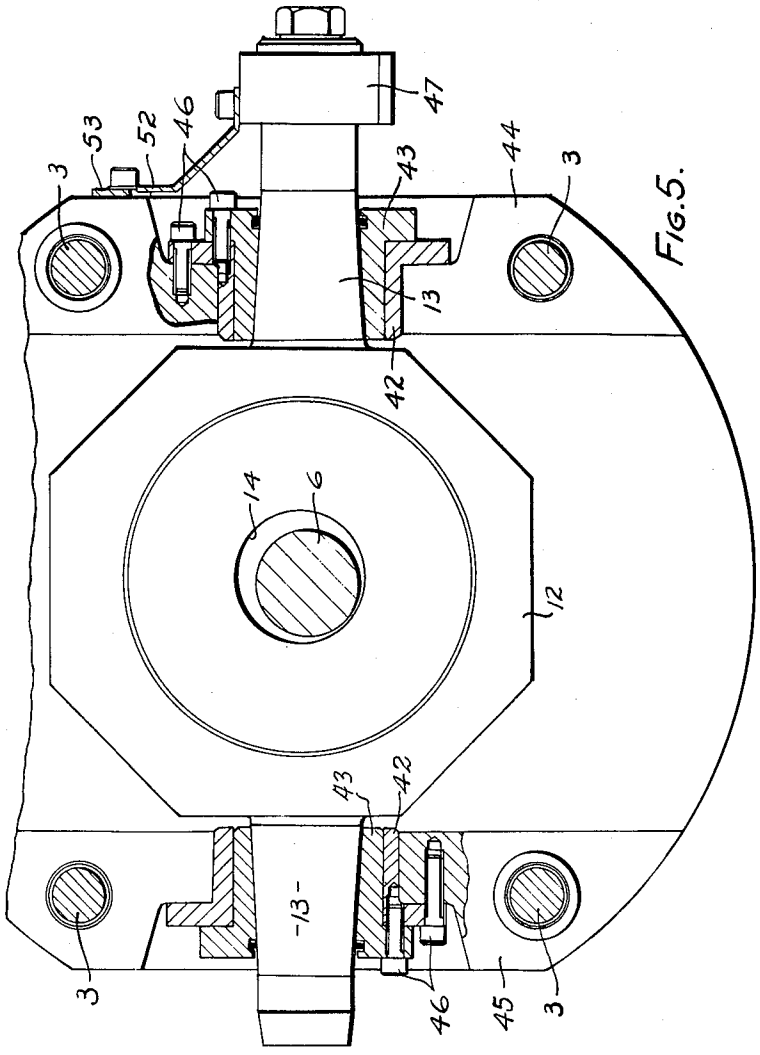


FIG. 5.

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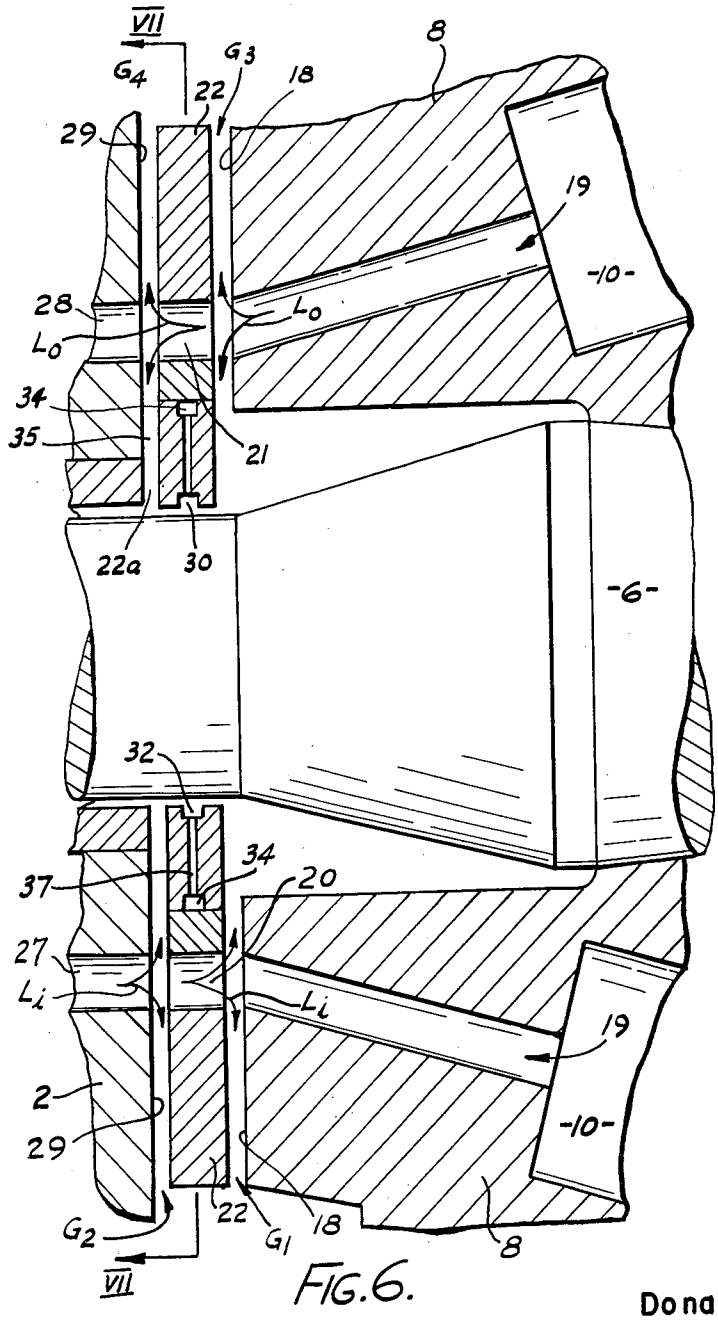


FIG. 6.

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HYDRAULIC MACHINES

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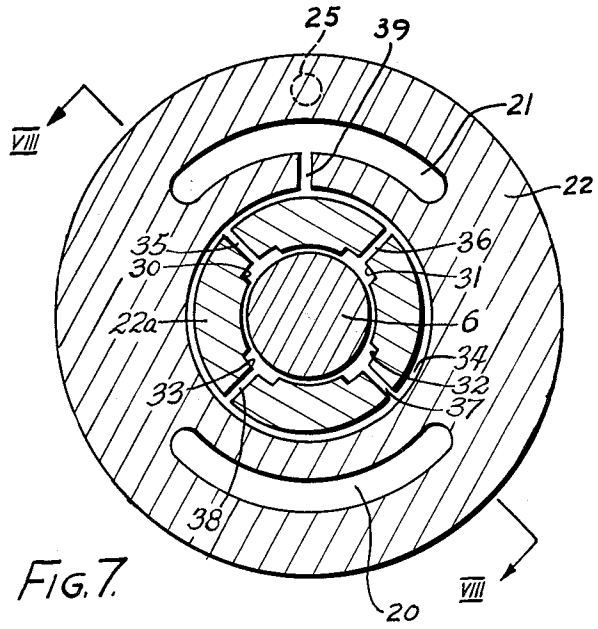


FIG. 7.

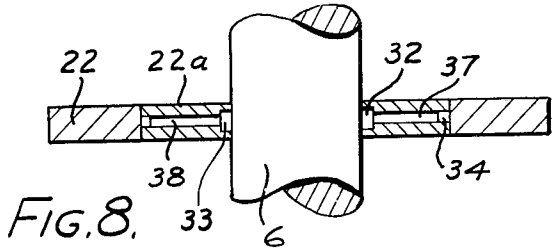


FIG. 8.

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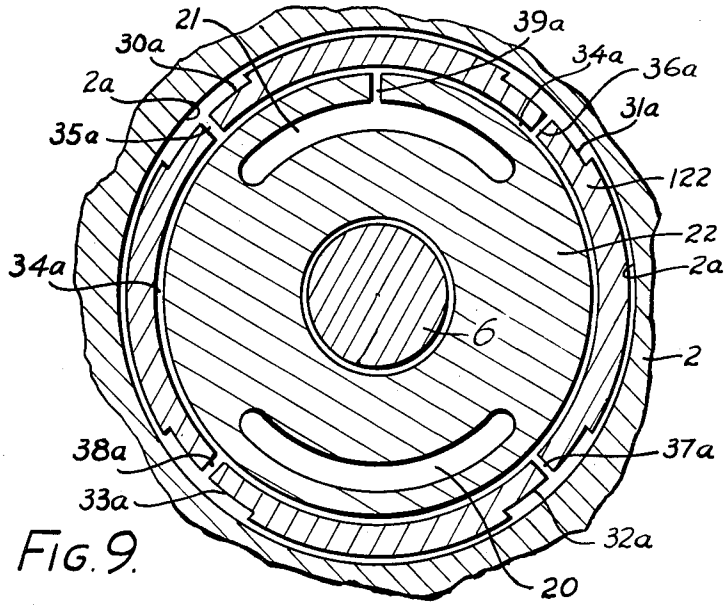


FIG. 9.

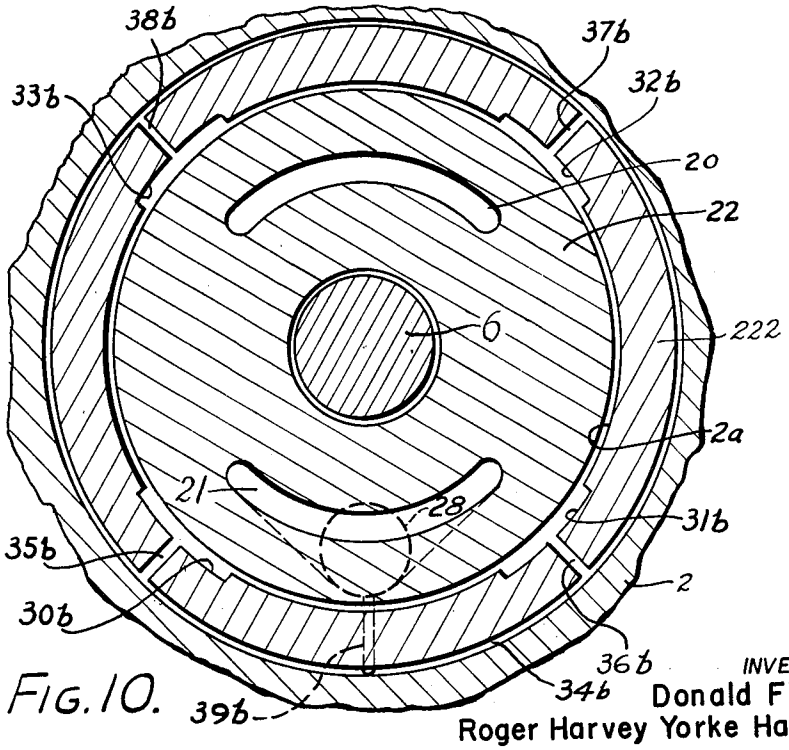


FIG. 10.

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United States Patent Office 3,089,427 Patented May 14, 1963

3,089,427

HYDRAULIC MACHINES

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Filed Apr. 14, 1960, Ser. No. 22,334
Claims priority, application Great Britain Apr. 16, 1959
13 Claims. (Cl. 103—162)

This invention relates to hydraulic machines, and particularly to swash plate type pumps and motors. Hitherto, in pumps of the swash plate type in which the cylinder block is rotated, considerable difficulty has frequently been encountered in developing sufficiently high pressures without risk of distortions or deflections of the cylinder block sufficient to cause seizure between it and the port or valve block or plate if the working clearances are kept to the very small values necessary to avoid undue leakage. The usual steps taken to combat these difficulties have hitherto been to increase the overall diameter of the cylinder block so as to impart a higher resistance to distortion, and to employ a flexible drive shaft or spherical seatings between the cylinder block and the port or valve plate for maintaining the required small working tolerances. The former leads to high inertia, and correspondingly sluggish response to signals, and the latter to mechanical complication.

Furthermore, it has usually been found necessary—in order to maintain the necessary fine working tolerances and to reduce friction losses—to support the cylinder block in comparatively large ball or roller bearings.

It is an object of the present invention to avoid or reduce to a minimum the difficulties hitherto experienced in the design and operation of swash plate pumps with rotary cylinder blocks without the disadvantages of high inertia and friction or substantial risk of seizure at high output pressures.

A further object is to provide a construction which embodies a high degree of automatic hydrostatic balance between the mechanical and oil lubricant pressures at the working surfaces of the port plate so that friction losses are kept very low.

By arranging the parts according to the present invention so as to make use of the principle of hydrostatic balance by means of oil films between the various working surfaces, it has been found possible to produce a design of pump which is not only of relatively small overall dimensions with low frictional drag, but also one which embodies simple and inexpensive plain journal bearings for supporting the shaft of the cylinder block.

A principal feature of the present invention is the interposition, between the working face of the cylinder block and a ported mounting block or end plate of the pump carrying the inlet and delivery connections, of an annular port plate concentric with the cylinder block shaft and whose opposite faces are at all times substantially supported on oil films the pressure of which is a function of the mechanical load imposed on the plate by the compression of the oil or other hydraulic working fluid in the cylinder block.

The port plate has a limited degree of radial "float" which is controlled by radially opposed hydrostatic pressures applied between the port plate and a relatively stable radially locating surface on the machine so as to vary differentially with variation of the clearance between the port plate and the said reference surface.

Conveniently, the opposed hydrostatic forces governing the radial float of the port plate act between the inner periphery of the annular plate and the radially stable surface of the shaft, the said periphery of the plate being formed with oil retaining cavities or pockets symmetrically arranged and connected to a source of oil under pressure so as to produce radially opposed stabilising oil pressures between the port plate and the shaft the difference between which at any instant is a function of any radial force tending to displace the floating port plate radially and cause metal to metal contact between it and the shaft.

If preferred, the radially opposed stabilising oil pressures on the port plate may be applied at the periphery of the port plate by locating the plate in a recess in the adjacent surface of the frame or end cover of the machine, and forming the pressure oil cavities or pockets either in the circumferential edge of the port plate or in the surrounding wall of the recess.

All the oil cavities or pockets may conveniently be interconnected by a common channel or groove which in turn is fed with oil under pressure from the high pressure port of the machine.

The necessary radial floating action of the port plate depends on the achievement of a significant reduction in oil pressure in a cavity or pocket whenever the clearance between the port-plate and its relatively stable radially locating surface increases in the vicinity of the cavity or pocket. Thus, each pocket must be connected to the source of oil under pressure through a narrow duct or constriction across which a rapid pressure drop is built up as the rate of flow tends to rise.

A hydraulic swash plate pump according to the present invention is capable of high power outputs—e.g. 200 H.P.—at speeds which are variable over a wide range and in which rapid response can be obtained to input control signals. The reductions possible in bearing surface areas reduce frictional loss, and hence friction heat, in the working parts.

A preferred embodiment of the invention will now be particularly described, by way of example only, with reference to the accompanying drawings in which:

FIGURE 1 is an axial cross section of the complete pump assembly;

FIGURE 2 is an elevation of the discharge end of the pump;

FIGURE 3 is a fragmentary section on the line III—III of FIGURE 2;

FIGURE 4 is a fgragmentary side elevation as seen on the arrow IV in FIGURE 2;

FIGURE 5 is a section on the line V—V of FIGURE 4;

FIGURE 6 is a diagrammatic sectional view of the port plate arrangement;

FIGURE 7 is a cross section on the line VII—VII of FIGURE 6;

FIGURE 8 is a sectional view on the line VIII—VIII of FIGURE 7 and

FIGURES 9 and 10 are sections similar to FIGURE 7 showing modified arrangements.

The pump illustrated consists of a main frame or body having front and ported back end plates 1, 2 clamped by four pillars 3 (FIGS. 2, 3 and 5). Each end plate 1, 2 carries a journal bearing 4, 5 respectively for a short rigid drive shaft 6. Adjacent the bearing 5 in the back end plate 2, the shaft 6 is formed with a locking taper section 7 on which is locked a cylinder block 8. This block is drawn up on the taper by a back-nut 9 on the shaft. The cylinder block 8 contains a number of cylinders 10 whose axes are mutually inclined inwards towards the back end plate 2. A piston 11 in each cylinder is reciprocable under the control of a normally fixed swash plate 12 carried on trunnions 13 (FIG. 5). The swash plate 12 has a central conical aperture 14 through which the shaft 6 passes, the dimensions of this aperture being sufficient to allow for adjustment of the angle of the swash plate to the shaft 6.

The working face of the swash plate is recessed to accommodate an annular bearing pad 15 and an annular

slipper plate **16**. The latter is free to rotate under the frictional drag of slippers **17** each of which is engaged with a respective piston **11**. For clarity of illustration in FIG. 1, only one cylinder **10**, piston **11** and slipper **17** is shown.

The back face **18** of the cylinder block **8** (see FIG. 6) is pierced by inlet/outlet ducts **19**, one to each cylinder **10**, the openings to which register successively with arcuate inlet and outlet ports **20**, **21** in a floating port plate **22**. This port plate is not anchored to the back end plate **2** except for angular orientation about the shaft **6** with respect to the swash plate **12**. This angular orientation is controlled by means of a threaded rod **23** which is slidable in a bore **24** in the back end plate **2** and which engages, at its inner end, a locating peg **25** fastened to the port plate **22** and projecting into the bore **24**. The rod **23** is axially adjustable by means of a captive capstan nut **26**. The ports **20**, **21** communicate with an external circuit by way of flared ducts **27**, **28** respectively (see FIGS. 1 and 2) and inlet and outlet sockets **27***a*, **28***a* in the back end plate **2**. The ports **27**, **28** open through a bearing surface **29** on the back end plate to register with the arcuate ports **20**, **21**, respectively in the port plate **22**.

The port plate **22** forms a kind of washer between the mutually opposed faces **18** of the cylinder block **8** and **29** of the back end plate **2**, and has a limited freedom to float radially between the two. This radial floating action is hydrostatically controlled by means of internal ducts and cavities.

As shown best in FIGS. 7 and 8, the port plate **22** has a central insert or hub **22***a* which fits over the shaft **6** with slight radial clearance or "float." Around the internal periphery of the hub **22***a* are four equally spaced oil pockets **30**, **31**, **32**, **33**. Each pocket communicates with an outer peripheral groove **34** around the hub **22***a* by means of small-bore radial ducts **35**, **36**, **37**, **38** respectively, and this peripheral groove is in turn directly connected to the main arcuate outlet port **21** by means of a radial bleed passage **39**. Thus, each oil pocket **30–33** is fed with oil at pump delivery pressure, and since the pockets are equiangularly spaced around the internal periphery of the hub **22***a*, the radial forces exerted by the oil pressures therein tend to balance out and maintain the port plate **22** centralised with respect to the shaft **6**. In this floating position, there is a narrow leakage path for oil between the inner periphery of the hub **22***a* and the radially stable locating surface of the shaft **6**, and all leakage paths are of the same nominal size and hence are of equal flow resistance. There is, therefore, a continuous flow of oil through the bleed passage **39** into the groove **34**, and from this groove along each radial duct **35–38**. The pressure drops along these parallel paths are equal.

Let it now be assumed that the plate **22** is radially displaced with respect to the shaft **6** in the vertically downward direction in FIG. 7. The leakage paths from the pockets **30** and **31** are then restricted whilst those from the pockets **32** and **33** are increased. Less oil will then escape from the pockets **30** and **31**, and more will escape from the pockets **32** and **33**. Hence, the pressure in the pockets **30** and **31** will rise and that in the pockets **32** and **33** will fall, tending to restore the plate **22** to its centralised position in which friction with the shaft **6** is at a minimum.

Instead of being radially located on the shaft **6**, the port plate **22** may be located at its outer periphery by an annular wall on the back end plate **2** of the machine casing, which thus constitutes the relatively stable or reference surface. An increased radial clearance is then provided between the port plate **22** and the shaft **6** so as to avoid any risk of binding of the plate on the shaft. Alternative constructions of this modified arrangement are illustrated in FIGS. 9 and 10.

In FIG. 9, the port plate **22** is shown as having a peripheral groove **34***a* machined around its circumference which is connected to the pressure port **21** by an internal duct **39***a* drilled radially in the thickness of the plate. This groove is closed by a ring **122** shrunk onto the plate **22** to form a rim. The ring has four pockets **30***a*, **31***a*, **32***a* and **33***a* at 90° spacings around its circumference, these pockets being connected to the groove **34***a* by ducts **35***a*, **36***a*, **37***a* and **38***a* and corresponding in function to the pockets **30–33** described above with reference to FIGS. 7 and 8. The periphery of the rim **122** is located with small clearance within an annular wall **2***a* bounding the surface **29** on which the port plate **22** rests. The action of the oil in the pockets **30***a–33a* and the narrow clearance from the wall **2***a* results in the same kind of self-centering action as occurs at the shaft **6** in the arrangement of FIGS. 7 and 8.

FIG. 10 illustrates the same principle as FIG. 9 except that the pockets **30***b*, **31***b*, **32***b* and **33***b* are now formed in the wall **2***a* which constitutes the inner periphery of a ring **222** which is a force fit in a recess in the back end plate **2**. A circumferential groove **34***b* is machined around the outer circumference of the ring **222** and communicates with each pocket **30***b*, **31***b*, **32***b* and **33***b* by ducts **35***b*, **36***b*, **37***b* and **38***b*. It also registers with a supply duct **39***b* leading from the high pressure port **28**. The action of this arrangement is identical with that of FIG. 9, as will be understood.

When the pump is working, half the total number of pistons **11** are being forced inwards against high oil pressure by the swash plate **12** whilst the other half are being forced outwards by low pressure oil. Since the thrust on each piston is taken through a slipper **17** on an inclined surface **16**, there will be a lateral component of reaction to each thrust which will be exerted on the respective cylinder wall and since these reactions are unequal over the two diametrically opposed arcs defined by the outlet and inlet ports **21**, **20** respectively, the cylinder block **8** experiences a tilting couple about the horizontal plane in FIG. 1. This couple tends to close the gaps at G_1, G_2 (FIG. 6) between the port plate **22** and the adjacent faces **18** of the cylinder block **8** and **29** of the back end plate **2** over an arc on the inlet side and at the same time to open the diametrically opposite gaps at G_3, G_4. This couple is resisted by the short, stout shaft **6**, and its magnitude is kept to a minimum by the reduction in overall diameter of the port face **18** of the cylinder block **8** which results from the inclination of the axes of the cylinder **10**, so that a frequent source of seizure or loss of output in pumps of this general kind, is accordingly minimized.

The floating port plate **22** itself also assists in mitigating possible trouble due to distortion of the cylinder block **8** under load. Although gaps are indicated in FIG. 6 at G_1–G_4, it will be understood that their dimensions are of the order of a few ten-thousandths of an inch. However, they exist physically and serve partly to cushion or accommodate cylinder block deflections and partly to combat distortion of the port plate **22** under fluctuating load conditions. The gaps are initially formed with the aid of capillary action.

Referring again to FIG. 6, the gaps G_1, G_2 are intersected by that part of the inlet circuit comprising the passage **27**, port **20**, and the relevant cylinder inlet/outlet ducts **19**. Similarly, the gaps G_3, G_4 are intersected by that part of the outlet circuit comprising the relevant cylinder inlet/outlet ducts **19**, port **21**, and outlet passage **28**. In each of these cases, the oil in the circuit is at a pressure above ambient pressure in the pump casing, the cylinders **10** being charged by oil at a positive inlet pressure from an auxiliary pump. Hence, leakage will tend to take place at both surfaces of the port plate **22**, as indicated by the arrows L_i in the gaps G_1, G_2, and L_o in the gaps G_3, G_4. Since the ports **20**, **21** are very large compared with the dimensions of the gaps G_1–G_4 the leakage pressure drop across the plate **22** is, for all practical purposes, zero, so that at all points around the plate the pressures on opposite faces are equal, although the pressures on either face of the plate at different points around the shaft **6**

vary from maximum pump delivery pressure to minimum inlet pressure. This range of variation may be from 2000 lbs./in.^{2} to 50 lbs./in.^{2}.

Provided that there is radial clearance or "float" between the port plate **22** and the shaft **6**, as described with reference to FIGS. 7 and 8, the port plate **22** floats axially irrespective of the actual value of pump delivery pressure, and the system is self-stabilising. Most of the couple tending to cause tilt of the cylinder block **8** due to uneven pressure loading is resisted by the short, rigid drive shaft **6**, and any residual deflection can be accommodated by compression of the leakage oil layers in the gaps G_1 and G_2. The aperture surface areas on opposite sides of the floating port plate **22** are proportioned so as to leave a relatively small residual unbalanced axial thrust on the shaft **6** which serves to overcome any tendency for the port plate **22** and cylinder block **8** to float axially in an uncontrolled manner leading to instability, and breakdown of the oil films. Due to the cyclic nature of the operation of successive pistons in each revolution of the cylinder block, a certain degree of "pumping" action is generated which tends to keep the films continuous, and this can, by careful design, be utilised to keep the mean unbalanced load between the plate **22** and the end plate **2** sufficiently low to prevent the port plate from binding on the end plate.

The combination of inclination of the cylinder axes and a fully floating port plate **22** materially reduces the frictional drag on the shaft **6**. The radial width of the annular face **18** of the cylinder block **8** determines the total force due to the pressure of leakage oil acting on the cylinder block. This force is balanced by the force acting on the area of the end wall of each pumping cylinder **10** around the outlet duct **19**.

The smaller the pitch circle diameter of the ports **20**, **21**, the less is the effective radius at which leakage oil in the gaps G_1 and G_3 of FIG. 6 is sheared, and the lower is the power loss absorbed thereby. On the other hand, if the amount of leakage at these gaps is to be kept within practicable limits, the difference in radius of this pitch circle and the outer circumference of the face **18** and port plate **22** must be as large as possible in order to give a long leakage path of high resistance.

All these factors influence the design of the cylinder block **8** and port plate **22**, and determine the angle of inclination of the cylinders **10** to the axis of the shaft **6**.

The provision for angular adjustment of the port plate **22** enables smoother operation of the pump to be obtained than is possible with conventional swash plate pumps in which no such adjustment is possible. The reason for this is that, since the pistons **11** cannot suck oil into their cylinders **10**, charging of the cylinders through the inlet port **20** must be effected under positive pressure. This must necessarily be low in comparison with the delivery pressure, so that at T.D.C. of the piston **11** (i.e. the completion of the charge stroke), the cylinder **10** contains oil at low pressure. If now the common inlet/outlet duct **19** is opened to the delivery port **21**, a pressure surge will be experienced in the cylinder due to the much higher pressure in the load circuit, and a series of shock waves would be set up in the system each time a piston commenced its delivery stroke. Hence, a small angular lag must be provided between T.D.C. and the opening of the duct **19** to the delivery port **21** to allow the oil to be pre-compressed in the cylinder **10** to at least approximately line pressure. The correct setting of the angular adjustment rod **23** which controls the port plate setting may be obtained by trial and error—as by means of the captive capstan nut **26**—or may be determined by a servo mechanism responsive to load on the pump. In this latter event, the fully floating feature of the port plate **22** considerably reduces the effort required by the servo motor, so that a more rapid response to changes in load can be achieved, with consequent improvement in the smoothness of running.

The swash plate **12**, as already noted, is mounted on trunnions **13**, and these are carried in bushes **42**, **43** (FIG. 5) mounted in side walls **44**, **45** of the pump casing. The bushes **42**, **43** are locked in position by means of cap screws **46**.

Since the swash plate **12** takes the whole of the reaction of each piston **11**, it will experience both the cyclic variations in load on each piston and also any shocks due to mis-timing of the outlet or delivery port **21**. Vibrations will, therefore, be set up in the swash plate **12** which are preferably absorbed by some convenient damping means (not shown) and operating either internally (i.e. within the structure of the swash plate assembly) or externally on any of the conventional principles, such as a dashpot, boundary lubrication, viscous or dry friction, inertia, or a combination thereof.

The angle of tilt of the swash plate **12** is adjusted by means of an arm **47**(FIG. 4) locked on the projecting end of one trunnion **13**, the extremity of which is clamped between a pair of adjustment screws **48** threaded through brackets **49** bolted on the side wall **42**. Each screw **48** has a knurled head **50** and carries a knurled locking nut **51**. A pointer **52** projecting from the boss of the arm **47** registers with a scale **53** graduated in degrees of tilt of the swash plate **12**.

The working face of the swash plate is constituted by the freely rotatable circular slipper plate **16** which beds against the bearing pad **15** in a circular recess in the swash plate. The bearing pad **15** is locked against rotation by means of a dowel. Thus, the friction between the piston slippers **17** and the slipper plate **16** causes the latter to tend to follow the slippers **17** around the shaft **6**, whilst the eccentricity of the plate due to the angle of tilt of the swash plate **12** causes the slippers **17** to trace a path over the working surface of the slipper plate which is not of constant configuration. Thus wear of the plate is distributed over an area greater than the annulus traced by a single slipper **17** if the plate **16** were stationary.

In a practical hydraulic transmission system embodying the invention, the speed ratio between the drive for the pump and the motor operated by the pump is infinitely variable over a relatively wide range by adjustment of the angle of tilt of the swash plate **12**. The arm **47** (FIGS. 4 and 5) in such an arrangement is linked to a speed control instead of being clamped in any given position of angular adjustment by the screws **48**. This speed control may be either manual or automatic in response to some external variable associated with the motor output, according to the requirements of the case.

It is to be understood that, with appropriate design modifications, the pump described above may be used as a motor.

We claim:

1. A swash plate type hydraulic machine comprising a stationary swash plate, a rotatable cylinder block, a plurality of pistons reciprocable in cylinders in said cylinder block under control of the swash plate, ports in a flat face of the cylinder block communicating with the cylinders therein, a stationary port block having a flat face, inlet and outlet ports in said flat face of the port block registrable with the ports in the cylinder block, a port plate interposed beween the flat faces of the cylinder block and the port block and having apertures permitting the passage of fluid between the ports in the cylinder block and the ports in the port block, said port plate having opposite flat faces of substantially equal area and clearances being provided between said opposite faces and the flat faces of the cylinder block and the port block within the limits of which clearances the port plate is free to move axially, whereby pressure fluid from said ports maintains a fluid film on each flat face of the port plate giving substantially equal and opposite axial loadings, a radial locating member having an annular surface spaced with radial clearance

from an annular surface on said port plate within the limits of which clearance the port plate is free to move radially, a plurality of pockets disposed symmetrically around one of said annular surfaces and open toward the other annular surface and a constricted fluid-supply connection to each of said pockets.

2. A machine according to claim 1 wherein the fluid-supply connection comprises a passageway having constricted branches each leading to one of said pockets.

3. A swash plate type hydraulic machine comprising a stationary swash plate, a rotatable cylinder block, a plurality of pistons reciprocable in cylinders in said cylinder block under control of the swash plate, ports in a flat face of the cylinder block communicating with the cylinders therein, a stationary port block having a flat face, inlet and outlet ports in said flat face of the port block registrable with the ports in the cylinder block, a shaft carrying said cylinder block, an annular port plate disposed coaxially around said shaft and interposed between the flat faces of the cylinder block and the port block and having apertures therethrough registering with the ports in the port block, said port plate having opposite flat faces of substantially equal area and clearances being provided between said opposite faces and the flat faces of the cylinder block and the port block within the limits of which clearances the port plate is free to move axially, whereby pressure fluid from said ports maintains a fluid film on each flat face of the port plate giving substantially equal and opposite axial loadings thereto, a plurality of pockets spaced symmetrically around the inner periphery of said port plate and open toward the shaft, a radial leakage clearance being provided between the shaft and the port plate within the limits of which clearance the port plate is free to move radially, and a pressure-fluid connection from at least one of the aforesaid ports in the port block to said pockets through constrictions each associated with a separate pocket.

4. A swash plate hydraulic machine comprising a stationary swash plate, a rotatable cylinder block, a plurality of pistons reciprocable in cylinders in said cylinder block under control of the swash plate, ports in a flat face of the cylinder block communicating with the cylinders therein, a stationary port block having a flat face, inlet and outlet ports in said flat face of the port block registrable with the ports in the cylinder block, a port plate interposed between the flat faces of the cylinder block and the port block and having apertures permitting the passage of fluid between the ports in the cylinder block and the ports in the port block, said port plate having opposite flat faces of substantially equal area and clearances being provided between said opposite faces and the flat faces of the cylinder block and the port block within the limits of which clearances the port plate is free to move axially, whereby pressure fluid from said ports maintains a fluid film on each flat face of the port plate giving substantially equal and opposite axial loadings, a radial locating member having a circular recess into which the port plate fits with a radial clearance between the circumferential surfaces of the port plate and the recess, a plurality of pockets spaced symmetrically around one of said circumferential surfaces and open toward the other circumferential surface, and a constricted pressure-fluid connection to each of said pockets.

5. A machine as claimed in claim 4 wherein said pockets are disposed in the port plate.

6. A machine as claimed in claim 4 wherein said pockets are disposed in the radial locating member.

7. A swash plate type hydraulic machine comprising a stationary swash plate, a rotatable cylinder block, a plurality of pistons reciprocable in cylinders in said cylinder block under control of the swash plate, ports in a flat face of the cylinder block communicating with the cylinders therein, a stationary port block having a flat face, inlet and outlet ports in said flat face of the port block registrable with the ports in the cylinder block, a port plate interposed between the flat faces of the cylinder block and the port block and having apertures permitting the passage of fluid between the ports in the cylinder block and the ports in the port block, said port plate having opposite flat faces of substantially equal area and clearances being provided between said opposite faces and the flat faces of the cylinder block and the port block within the limits of which clearances the port plate is free to move axially, whereby pressure fluid from said ports maintains a fluid film on each flat face of the port plate giving substantially equal and opposite axial loadings, a radial locating member having an annular surface spaced with radial clearance from an annular surface of said port plate within the limits of which clearance the port plate is free to move radially, a plurality of pockets at equally spaced points around the annular surface of the port plate and opening toward the annular surface of the radial locating member, and a constricted pressure-fluid connection to each of said pockets from an aforesaid port.

8. A swash plate type hydraulic machine comprising a housing having an end plate with a flat surface, inlet and outlet ducts in said end plate for conveying hydraulic fluid, a rotatable shaft mounted in said housing on an axis normal to said flat surface, a swash plate mounted in said housing on said axis, a cylinder block fixed on said shaft and located between said end plate and said swash plate, cylinders in said block disposed around said axis, a port communicating with each of said cylinders and opening through a flat end face of said cylinder block opposite to said flat surface of the housing end plate for conveying said fluid to and from the cylinder, a reciprocable piston in each cylinder displaceable by said swash plate on relative rotation of said swash plate and said cylinder block, an annular port plate whose opposite sides have smooth flat surface of equal area, the port plate being mounted to float axially between said flat surface on the housing end plate and said flat end face of the cylinder block and to float radially with respect to said axis, pressure balancing means for limiting radial movement of the port plate, spaced arcuate transfer ports in said port plate respectively registering with said inlet and outlet ducts in the housing end plate and registrable in sequence with said ports in the cylinder block on rotation of the cylinder block, whereby fluid can be transferred by reciprocation of said pistons from the inlet duct, via said transfer ports and said ports and cylinders in the port block, to the outlet duct, said pressure balancing means comprising a plurality of pockets opening through an annular surface of the port plate and symmetrically disposed about said axis, a pressure-fluid connection to each pocket, a restriction in each pressure-fluid connection whereby the pressure in each pocket depends on the rate of escape of fluid therefrom, a radial locating member radially fixed relative to said axis and having a circular reference surface adjacent to and spaced from said annular surface of the port plate for controlling the rate of escape of fluid from said pockets in dependence on the spacing of said annular surface from the reference surface.

9. A machine according to claim 8 wherein said radial locating member comprises said shaft.

10. A machine according to claim 8 wherein said radial locating member comprises said casing and said circular reference surface is provided by the wall of a recess in the casing.

11. A machine according to claim 8 wherein said pressure-fluid connections lead to one of said arcuate transfer passages.

12. A machine according to claim 8 wherein the port plate is angularly adjustable relative to the swash plate.

13. A swash plate type hydraulic machine comprising a housing having an end plate with a flat surface, inlet and outlet ducts in said end plate for conveying hydraulic fluid, a rotatable shaft mounted in said housing on an axis normal to said flat surface, a swash plate mounted in said housing on said axis, a cylinder block fixed on said

shaft and located between said end plate and said swash plate, cylinders in said block disposed around said axis, a port communicating with each of said cylinders and opening through a flat end face of said cylinder block opposite to said flat surface of the housing end plate for conveying said fluid to and from the cylinder, a reciprocable piston in each cylinder displaceable by said swash plate on relative rotation of said swash plate and said cylinder block, an annular port plate whose opposite sides have smooth flat surfaces of equal area, the port plate being mounted to float axially between said flat surface on the housing end plate and said flat end face of the cylinder block and to float radially with respect to said axis, pressure balancing means for limiting radial movement of the port plate, and spaced arcuate transfer ports in said port plate respectively registering with said inlet and outlet ducts in the housing end plate and registrable in sequence with said ports in the cylinder block on rotation of the cylinder block, whereby fluid can be transferred by reciprocation of said pistons from the inlet duct, via said transfer ports and said ports and cylinders in the port block, to the outlet duct, said pressure balancing means comprising means for distributing pressure fluid in opposed relationship through an annular surface of the port plate symmetrically about said axis, a pressure-fluid connection to said means for distributing pressure fluid, restrictor means in said pressure-fluid connection for controlling the pressure in said means for distributing pressure depending on the rate of escape of fluid therefrom, a radial locating member radially fixed relative to said axis having a circular reference surface adjacent to and spaced from said annular surface of the port plate for controlling the rate of escape of fluid from said means for distributing pressure fluid in dependence on the spacing of said annular surface from the reference surface.

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